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Lubrication

A Technical Publication Devoted to the Selection and Use of Lubricants

THIS ISSUE

Bearing Lubrication
Oil Film Pressure

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LUBRICATION

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Bearing Lubrication

Oil Film Pressure

RESEARCH in the interests of developing data pertinent to plain bearing lubrication has been extensive, ever since the classic investigations of Osborne Reynolds. From a theoretical point of view his work, as well as the results of the various studies of such outstanding later investigators as Sommerfeld, and Harrison, is of extreme interest. This is all more or less involved, however, and frequently beyond the full comprehension of the average machine designer and practical operator. Yet, it is the latter who must keep machinery running, renew bearings where trouble had occurred, and make final choice as to his means and manner of lubrication.

In recognition of this, Howarth and Karelitz extended their studies along certain practical lines for the purpose of developing charts and data which the machine designer could actually use in ascertaining the characteristics of a proposed bearing prior to its construction, thereby to determine by use of these charts the relative effect of changes in bearing dimensions, or manner of lubrication. One must remember, however, that due to the assumptions as to distribution of pressure within the oil film, the accuracy of these charts will depend upon the agreement of the assumptions with actual operating conditions in the bearings.

With the purpose in mind of further developing the practical application of the theories already evolved, the Engineering Experimental Station of the Pennsylvania State College* has recently essayed to study oil film pressure within a complete journal bearing, lubricated from one end, as well as the running positions of the journal. This work has been sponsored in part by foundation of a research fellowship by The Texas Company.

Broadly speaking, the work of earlier investigators was taken as a basis, wherein calculation of theoretical pressure distribution, coefficient of friction and journal running position is made possible where the intensity of the load, journal speed and the dimensions of the journal and bearing are known.

DESCRIPTION OF THE APPARATUS

The machine used in this work was of unique and special design. Photographs of it are shown in Fig. 1, and drawings of the end elevation and longitudinal section are shown in Fig. 2. Fig. 3 shows a detail of the test bearing.

The machine consists of a shaft, 1, (Fig. 2), 2.495″ in diameter, which rests in two self aligning bearings, 2, and which carries the test bearing, 3. This test bearing is a heavy cylindrical steel shell lined with babbitt. The bore was carefully reamed and then accurately measured with a gauge which could be read to 0.0002″. Four diameters were measured at each axial position and the average of these measurements taken as the diameter at that point. These values are noted on the drawing of the bearing, Fig. 3. The length of the test surface of the bearing is 5″.

The shell is drilled with four radial holes, A, B, C, D, as shown in Fig. 3, which com-

*Bulletin No. 39, Oil Film Pressures in a Complete Bearing, by L. J. Bradford and L. J. Grunder. 1930.

municate with four axial holes, A′, B′, C′, D′, Fig. 2. Two thermometer wells are provided as shown at 27. The outside of the shell is accurately ground so as to be concentric with the bore.

The test shell fits into a cylindrical hole in a heavy, steel yoke, 17, Fig. 2, the hole being about 0.002″ larger in diameter than the outside diameter of the test bearing shell. When in place in the yoke the test shell is bolted to a nected to either of two of the axial holes. Thus either high or low pressures may be read accurately.

Bolted to the forging just described is a bronze ring, 11, divided into degrees, to enable the determination of the circumferential position of the bearing shell with respect to the yoke.

Four dial micrometers, 8, are also attached to this forging. The stems of the micrometers

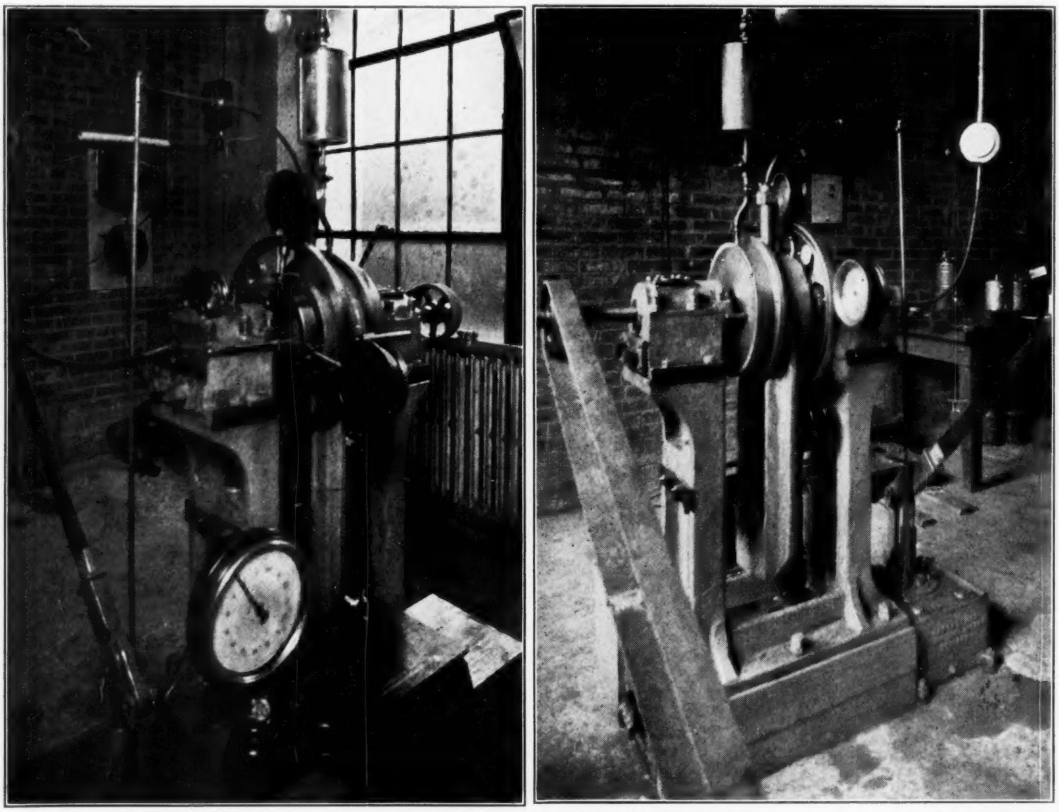


Figure 1
General view of the machine used in film pressure investigations at Pennsylvania State College.

counterweight, 15, contained in a space formed by shrinking a housing, 22, on the yoke. This counterweight does not touch either the shaft or the yoke.

A heavy steel forging, 4, is bolted to the test shell on the end opposite from the counterweight. Passages in it are connected to the axial holes in the test shell by means of soft copper connectors, 33. The passages lead through stop cocks and tubes, 7, to pressure gauges, 5 and 6.

There are four pressure gauges, two high pressure reading from zero to 1,000 lbs. per sq. in. and two low pressure reading from 30″ vacuum to 75 lb. per sq. in. The arrangement is such that by means of the stop cocks either a high or a low pressure gauge may be con-

rest against babbitt faced pins, 9, which are held against the shaft by means of springs, 10. The reading on the micrometer dial gives the displacement of the shaft with respect to the bearing at that point.

Oil for the test bearing is supplied by gravity through an oil filter, 21, and heater, under a head of 26 ft. to the space in which the counterweight is located. Thence it enters the bearing through the end, and after passing through falls out at the other end into a drip pan.

A hand operated pump, 24, supplies oil through a second filter, 25, to the space between the test shell and the yoke. Sufficient pressure can be developed by this pump to cause complete separation of the shell and yoke.

Due to the entire end of the test shell adjacent to the counterweight being acted upon by oil under supply pressure, while the opposite end is acted upon by atmospheric pressure only the bearing is acted upon by an unbalanced axial thrust of approximately 250 lbs. acting from the counterweight end. An equal and opposite thrust acts on the counterweight housing. These forces are cared for by a pair of rigid stops, bearing against the housing, and a pair of rollers, bearing against the forging attached to the bearing. The rollers are carried on the ends of a pair of rods, 31, which pass through close fitting sleeves. The outer ends are cone pointed and rest in cup pointed set screws, attached to an equalizer plate. The sleeves may be withdrawn from their supports, and the rods left free to pivot about the set screws. When this is done the rollers will prevent axial motion of the bearing, but will exert no tangential force upon it, and will, therefore, not affect the friction reading.

A rod, 13, may be bolted to the graduated ring, 11, and the free end rested on the platform of a scales, 14. Any tendency of the bearing to rotate will produce a force in this rod which may be weighed. This is used in determining the amount of friction set up.

At its lower end the yoke carries a large set screw, 16, which bears against a set of double knife edges. A heavy steel bar, 18, rests on this knife edge set. The ends of the bar rest against a pair of double knife edges, which in turn rest against set screws, supported in the frame of the machine. The center to center distance of these latter set screws is $15\frac{1}{2}''$. Resting on the bar is a plate, 20, which is point supported and which carries a dial micrometer, 19. When set screw, 16, is tightened the bar, 18, is deflected upward and a downward load is applied to the yoke and through it to the bearing. This load is proportional to the deflection of the bar, which can be measured by means of micrometer, 19.

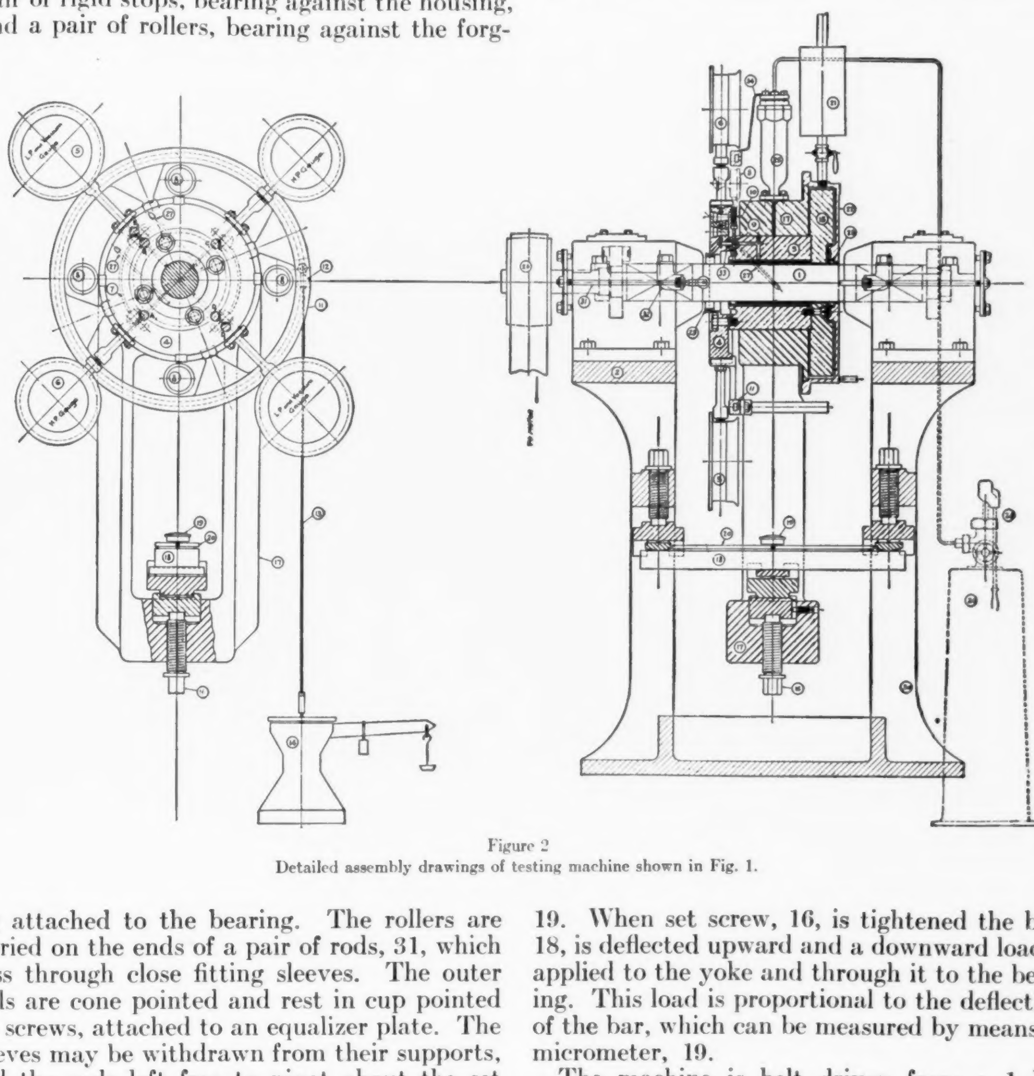


Figure 2
Detailed assembly drawings of testing machine shown in Fig. 1.

The machine is belt driven from a 1 hp. shunt wound D.C. motor.

Gauge for Measuring Bearing

A special gauge was made for measuring the internal diameter of the bearing. It is shown in Fig. 4. It consists of two steel bars, one of which is shown at 1. The other is concealed in a groove in the under side of the support bar, 3. These two bars are joined together at one end by a piece of clock spring, 7, which furnishes a flexible but inextensible connection,

entirely free from the lost motion which would be present in a pin joint. Attached to these bars are two spurs, 2. A dial micrometer, 4, is attached to one of the bars at the end, its

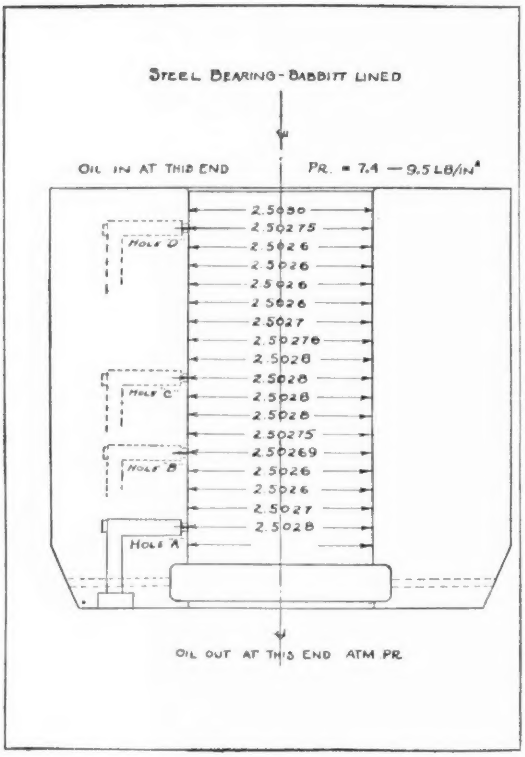


Figure 3
Longitudinal section of test shell, showing relative location of radial holes.

stem bearing against the other bar. It thus indicates displacement of one bar with respect to the other. This mechanism constitutes the gauge proper. It rests on a pivot, 5, in a suspender bar, 6, which, in turn is screwed to support bar, 3. The spurs, 2, pass through holes in the support bar, 3, and suspender bar, 6, and are free to move with respect to these latter bars. The proportions of the gauge are such that the motion recorded by the micrometer is about five times the motion of the spurs. The support and suspender bars support the bearing under test and the gauge. They take no part in the actual measurement of bearing diameter.

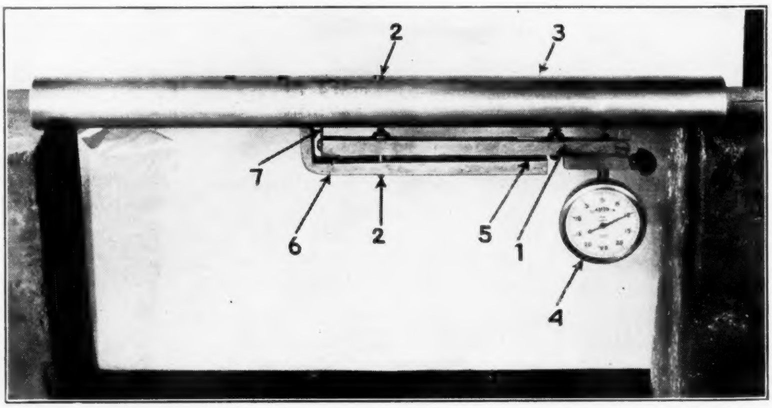


Figure 4
Details of gauge for measuring internal diameter of test shell.

In using the gauge to determine the diameters of the bearing the support bar was passed through the bearing and the ends rested on substantial supports. The bearing rested on the support bar, with the spurs of the gauge bearing lightly against the inner surface of the bearing. The bearing was then rocked back and forth on the support bar and the dial micrometer readings noted. The largest value was accepted as being the diameter at that point. Four different diameters were measured and the average taken as the bearing diameter at that station. Stations were taken at ¼″ axial intervals.

Calibration of Instruments

The gauge for measuring the bearing, the load bar, the pressure gauges, as well as thermometers were suitably calibrated prior to actual usage. Texaco Regal Oil "C" was used extensively in the subsequent operations of the testing machine.

METHOD OF CONDUCTING WORK

As already stated, the primary objects of the test were:

(a) To determine the distribution of pressure within the oil film when the shaft was completely separated from the bearing by the film.

(b) To determine the running position of the shaft with respect to the bearing.

The first step was to determine the load at which metallic contact between the shaft and bearing occurred. This was done by placing a 3-volt flashlight in a circuit which included the oil film, and so insulating the machine that a complete circuit was impossible unless the shaft and bearing were in electrical contact. It was found that the light began to glow when the load reached a value of 415 to 425 lbs. per sq. in. of projected area, for a speed of 500 r.p.m. Since the light did not glow at pressures less than these it was concluded that the shaft

and bearing were completely separated when the pressure was less than 415 lbs. per sq. in.

Two values of speed were used, 500 and 750 r.p.m.

The temperature was controlled by opening or closing windows near the machine, and could be held fairly constant at between 105 degrees Fahr., and 113 degrees Fahr. This seems to be the normal running temperature for the machine. Several attempts were made to run at considerably higher temperatures, securing these by heating the oil entering the machine. These attempts proved futile. The quantity of oil passing through the machine was so small, 2.5 cu. in. per minute, and the mass of the machine so large that even with the oil entering the housing at 200 degrees Fahr. and a room temperature of 90 degrees Fahr., the temperature of the bearing shell could not be raised above 120 degrees Fahr.

In making a run the machine was first operated until the temperature of the bearing remained constant for about half an hour. This took, usually, about three hours. The cock connecting one of the holes to one of the gauges was then opened and, after allowing the pressure in the gauge to become steady, the value was read and the position of the hole ascertained by means of the graduated ring, 11, Fig. 2. Oil was then forced between the yoke and shell by means of the hand operated pump until the two were completely separated. The bearing was then turned a few degrees with respect to the yoke and the pressure again noted. This operation was repeated until pressures had been observed for a full 360 degrees, high and low pressures being read from the respective gauges.

The speed was determined at frequent intervals by means of a stop watch and revolution counter, and was carefully maintained at the number desired, either 500 or 750 r.p.m. The temperature of the shell was also checked from time to time and was kept, as nearly as possible, at the initial value. Due to the large mass of the machine and the uniformity of the room temperature the variations in bearing temperature were slight.

It was found highly important to allow plenty of time for the pressure in the gauges to reach equilibrium. This was especially true at light loads, and in regions of low pressure. Under these conditions the pressures were not only slow to change, but were apt to fluctuate considerably, especially those recorded from the holes adjacent to the outlet end of the bearing. It is believed that these fluctuations were due to the film having broken, and air being drawn into the bearing in the region of negative pressure on the outlet end.

After completing a transverse of 360 degrees with one hole, the same thing was done with another, and this repeated for the third and fourth holes.

The pressures obtained in this manner were plotted against angular position of the hole, measured in the direction of rotation from a line of right angles to the load on the "on" side of the bearing. Figs. 5 to 8 show these curves for a speed of 500 r.p.m. using Texaco Regal Oil "C". Figs. 9 to 12 are for a speed of 750 r.p.m. with the same oil.

The shaft position with respect to the bearing was found by setting each dial micrometer at a convenient reading when it was at zero degrees to the line of action of the load. All micrometers were set at the same reading. The readings of the micrometers were noted at the same time the pressures were read.

The shaft position was obtained by drawing a circle of any convenient size and plotting the various micrometer readings radially from this circle, to any convenient scale. A radius and center were then found by trial, such that a circle could be drawn through the plotted points. The distance between the center of this circle and that of the original circle gave the eccentricity of the shaft with respect to the bearing, and a line drawn through the two centers passed through the point of closest approach. Obviously some judgment had to be used in drawing this circle.

These diagrams are shown on Figs. 5 to 12 and are drawn for the conditions of load and speed indicated.

Attempts to secure the friction were unsatisfactory. In order to free the test shell from the yoke completely when measuring the friction it is necessary to force oil between the two. Before this oil is introduced the shell and yoke are in contact along the upper surface of the shell. When oil is introduced through the filter from the hand pump, 24, the yoke is lifted, the amount of the lift ranging up to the value of the diametral clearance between the shell and the yoke. When this lifting of the yoke occurs it causes a corresponding increase in the deflection of the load bar and consequent increase in the load applied to the bearing. Since the hand pump is of the single cylinder, single acting type its discharge varies greatly and, in consequence, the thickness of the film separating the yoke and the shell varies, causing, as stated, a considerable variation in the load applied to the bearing. The result is that friction values are rather unreliable. The mean values obtained are shown on Figs. 5 to 12.

It was noted on several occasions when the pressure gauges were removed that the oil flowing from the connections contained minute bubbles of air. It was concluded that this air was drawn into the bearing in the low pressure

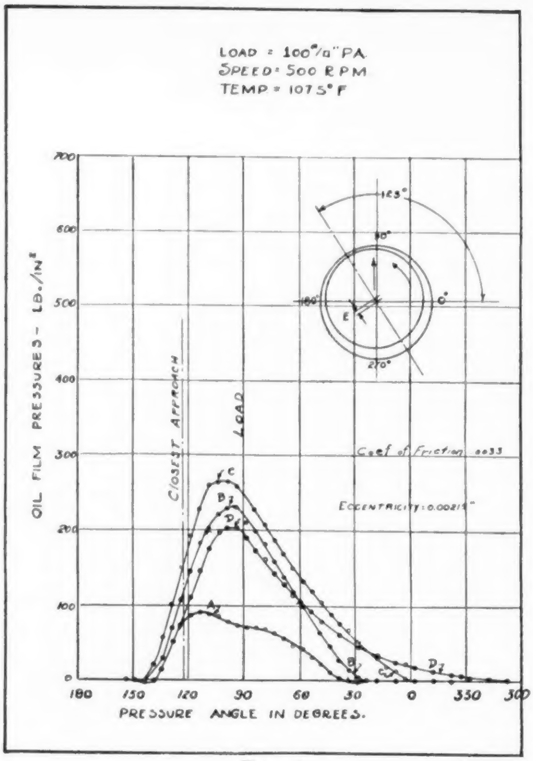
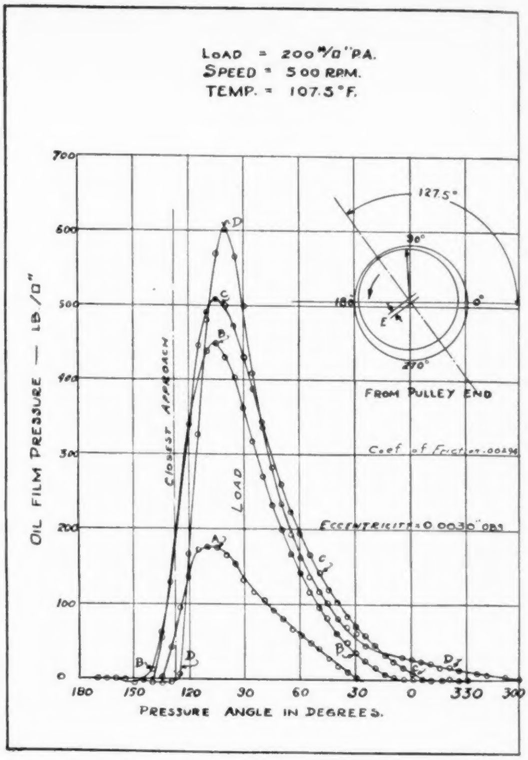


Figure 5



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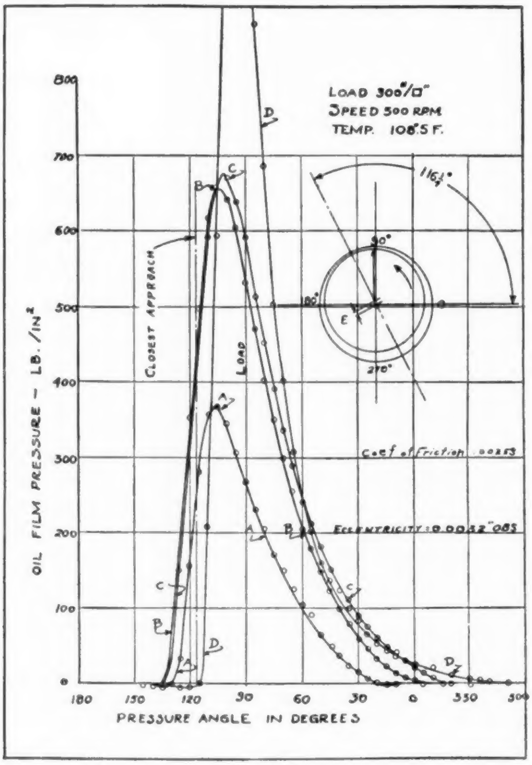

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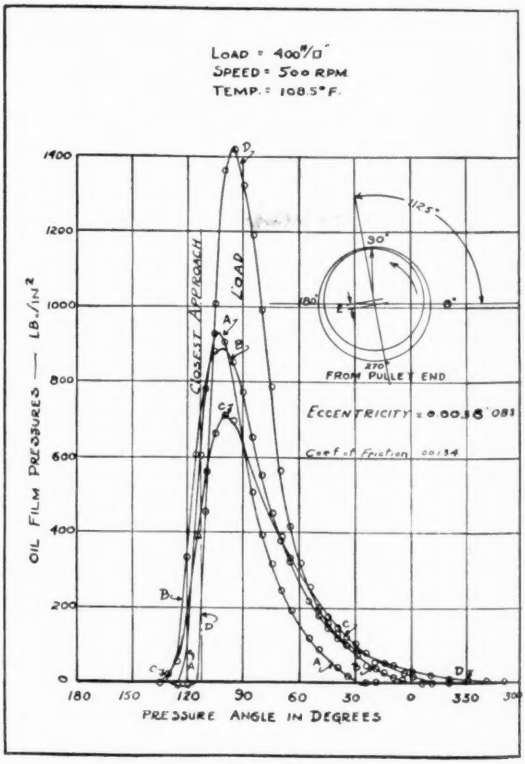

Figure 8

Pressure curves for tests on Texaco Regal Oil "C" at 500 r.p.m. Fig. 5 shows curves at 100 pounds pressure per square inch; Fig. 6 shows 200 pounds pressure; Fig. 7 shows 300 pounds pressure, and Fig. 8 is for 400 pounds pressure.

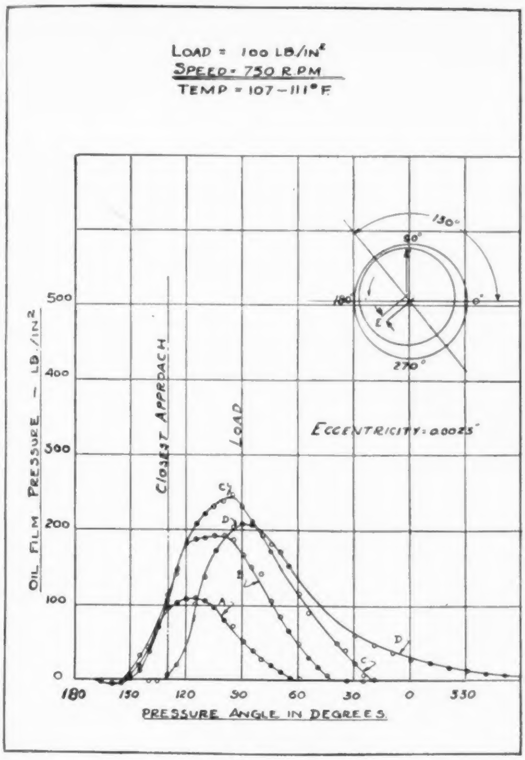

Figure 9

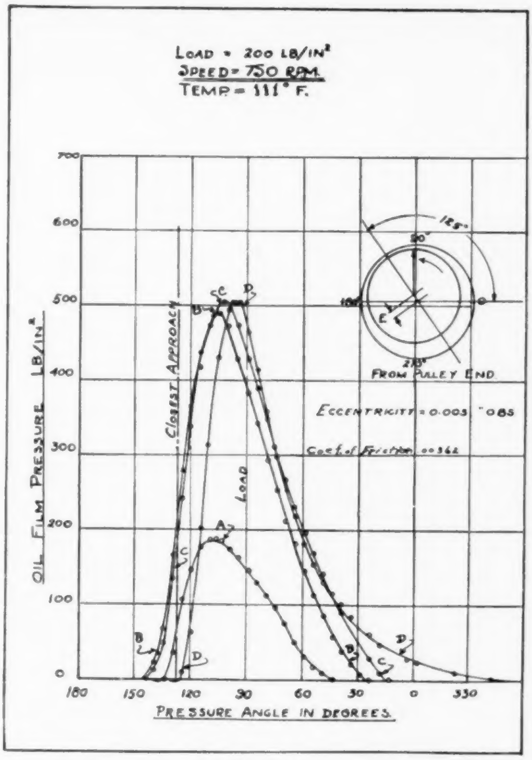

Figure 10

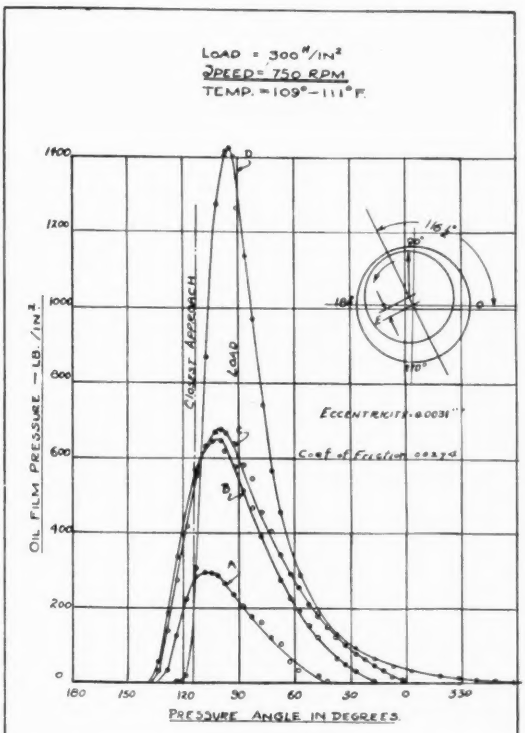

Figure 11

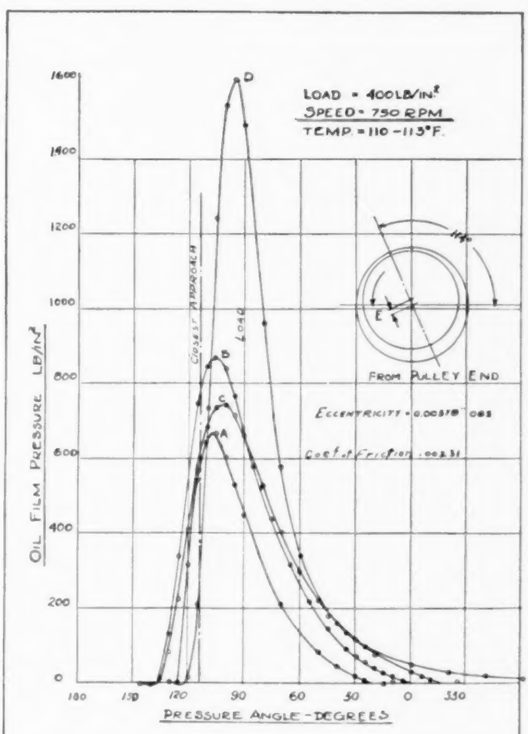

Figure 12

Pressure curves and shaft position for Texaco Regal Oil "C" at 750 r.p.m. Fig. 9 shows curves at 100 pounds pressure per square inch. Fig. 10 shows 200 pounds pressure. Fig. 11 is for 300 pounds pressure, and Fig. 12 shows 400 pounds pressure.

region on the outlet end, became mixed with the oil and was carried around to the region in which the oil film sustained a positive pressure.

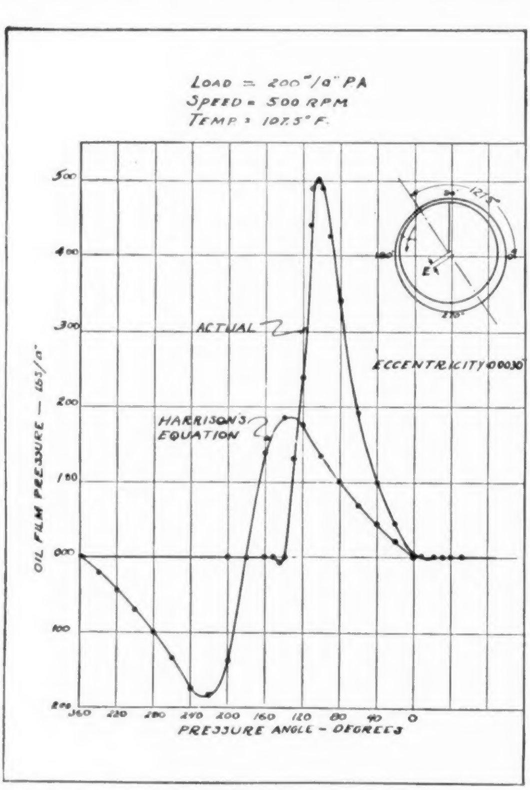

Figure 13

Showing pressure curves for 200 pounds per square inch at a speed of 500 r.p.m. Note that the curve has first been plotted as actually observed and then as calculated from Harrison's equation for a complete bearing.

In this way air found its way into the bearing as far as the center hole. In order to determine whether this was the case a stuffing box was made up and clamped on the outlet end of the bearing. This was arranged so that it did not fit tightly enough against the shaft to prevent the latter from assuming any position it desired with respect to the bearing. It was just tight enough to restrict the outflow of oil sufficiently to cause it to fill the groove at the end of the bearing (See Fig. 3). With this groove filled with oil and the opposite end of the bearing completely covered with oil in the housing the inflow of air into the bearing became impossible.

RESULTS OBTAINED

The first and most striking result was the indication that the pressure distribution within the oil film differed widely from that indicated by the classical theory. The latter indicates that the positive and negative pressures should be equal. This is manifestly impossible since negative pressures below zero absolute cannot be produced. The oil film must, therefore, break when the negative pressure reaches some value dependent upon factors as yet undetermined. The load carrying effect of the negative loop is lost and must be compensated for by an increase in the values obtained in the positive pressure loop. This can be secured only by an increase in the eccentricity of the shaft with respect to the bearing. This is exactly what happens, and is shown clearly on the curves of oil film pressure, and shaft eccentricity, Figs. 5 to 12. For the purpose of comparison Fig. 13 shows the theoretical pressure distribution in a complete bearing for a load of 200 lbs. per sq. in. and a speed of 500 r.p.m. superposed on the actual pressure distribution for these conditions obtained at the center of the bearing, hole C. It would appear that a complete bearing be-

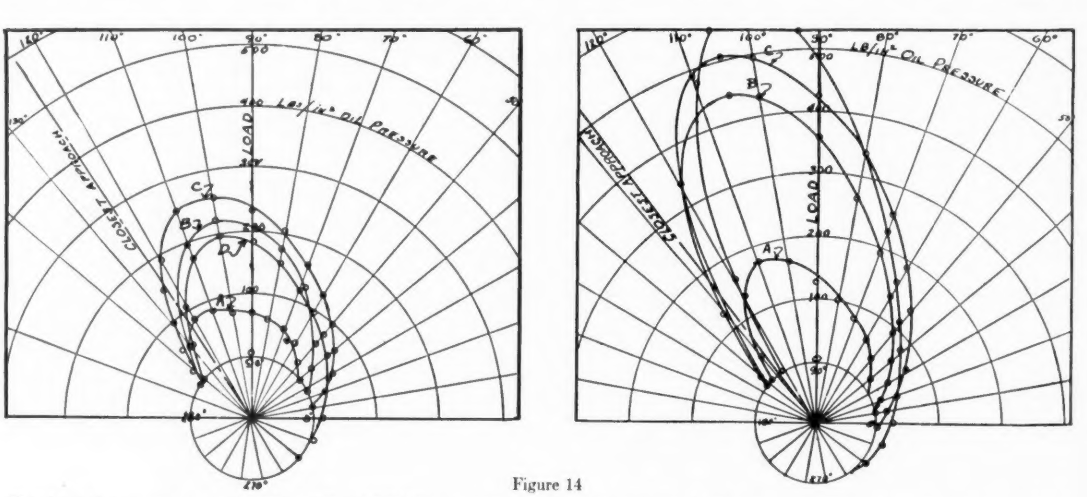

Figure 14

Showing polar pressure curves for Texaco Regal Oil "C" at a speed of 500 r.p.m. at 100 and 200 pounds pressure per square inch respectively.

haves as a partial bearing, although the equivalent partial bearing appears to be different for each set of conditions. Fig. 14 shows the same values as Fig. 5, plotted on polar coordinate paper in order to show more clearly the portion of the bearing which is effective in sustaining the load.

The data shown in Figs. 6 to 12 has been similarly plotted in the original paper, for the various conditions of load and speed. This does not show that the pressure distribution predicted by Harrison will not obtain when the conditions he assumes are present. It does show that in a cylindrical sleeve bearing, of the type dealt with in this investigation, supplied with oil from one end, the assumed conditions are not present, and that serious errors in both eccentricity and pressure distribution will be incurred by applying Harrison's equation for a full bearing to the type of bearing dealt with here.

Figures 15 to 22 show the longitudinal distribution of pressure, and were obtained by cross plotting Figs. 5 to 12. The bearing angle at which each curve was plotted is indicated thereon.

It will be noticed that the longitudinal distribution of pressure is not symmetrical about the center of the bearing. This was due to the center of pressure of the yoke against the bearing shell not being directly over the center of the active portion of the bearing, but 3/16″ closer to the inlet end of the bearing. The adjustments available on the machine were not sufficient to permit the resultant of the pressure of the yoke against the shell being made to pass through the center of the active portion of the bearing, and an unsymmetrical distribution of pressure within the oil film inevitably resulted.

The shape of the longitudinal pressure distribution curves is roughly parabolic at a load of 100 lbs. per sq. in. As the load is increased these curves first flatten at the top and then become saddle shaped. This is due to shaft deflection within the bearing. The shaft deflection causes the shaft to run closer to the bearing at the ends than at the middle, which means that the eccentricity of the shaft is greater at the ends of the bearing than it is at the middle. The pressure built up within the oil film varies directly as a function of the eccentricity, other things being equal, and, therefore the longitudinal pressure distribution found in the higher loads is to be expected when shaft deflection within the bearing length becomes appreciable. Assuming that the load was uniformly distributed over the length of the bearing, and that the shaft was supported on knife edges at the center lines of the supporting bearings the maximum deflections occurring within the active portion of the bearing were:

Load lb. per sq. in.	Deflection inches
100	0.000214
200	0.000428
300	0.000642
400	0.000856

The shaft displacements differed from those indicated by the classical theory for a complete bearing both in amount and direction. The eccentricity was considerably greater than theory indicates for a full bearing, and the line joining the center of the shaft with that of the bearing made an angle of between 12 degrees and 40 degrees with the line of action of the load. Theoretically this angle should have been 90 degrees in all cases.

Sealing the outlet end of the bearing with oil so that air could not enter the clearance space caused marked changes in the oil film pressures and shaft displacement. Only one run was made, that being at 750 r.p.m. and a load of 200 lb. per sq. in. of projected area. The results of this run are shown in Fig. 23, and are summarized below:

	Sealed end	Unsealed end
Eccentricity	0.0029″	0.0030″
Maximum Positive Pressure	650 lbs./sq. in.	500 lbs./sq. in.
Maximum Negative Pressure	15″ mercury	9″ mercury
Angle between line of load and line of shaft and bearing	50 degrees	35 degrees

It would appear from this that the capacity of the bearing can be materially increased by preventing the ingress of air in the region of negative oil film pressure. Further work is needed, however, to establish this fact conclusively.

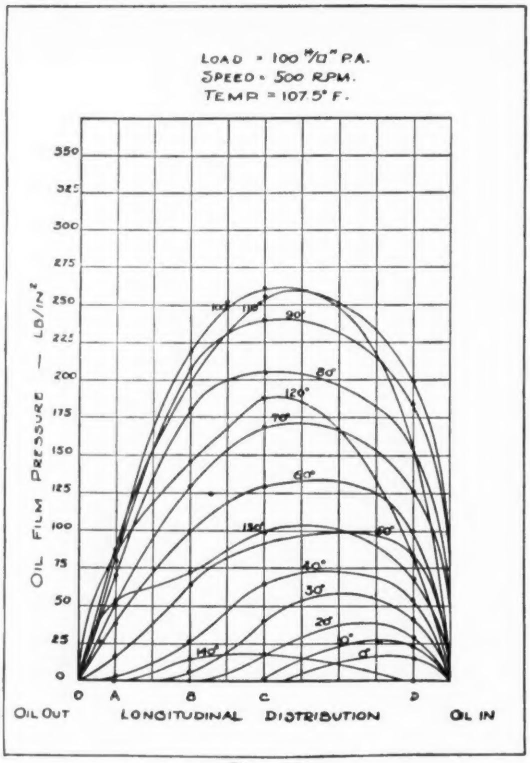

Figure 15

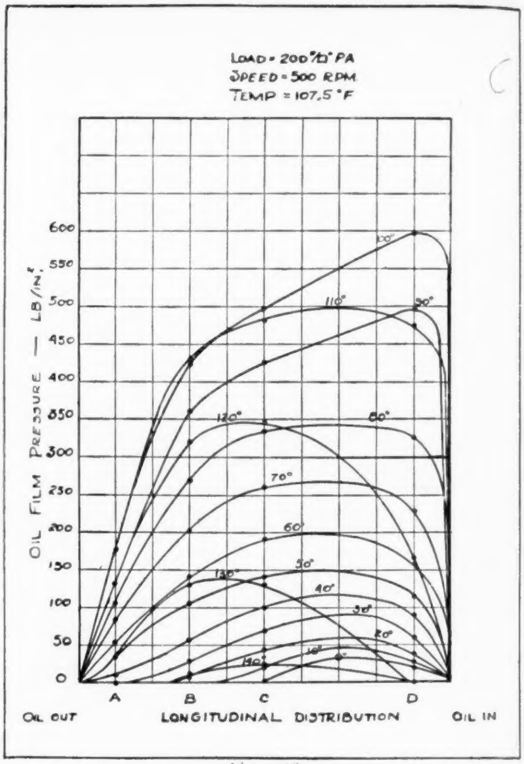

Figure 16

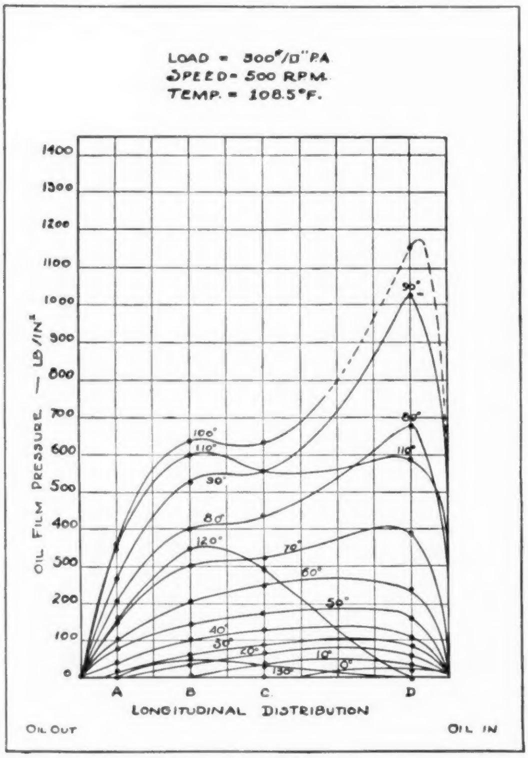

Figure 17

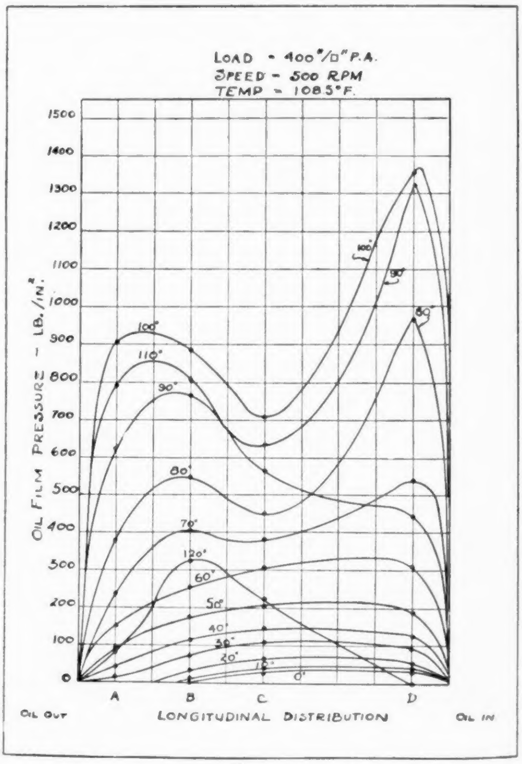

Figure 18

Showing longitudinal pressure curves for various angles, measured in the direction of rotation from a line at right angles to the load on the "on" side, for Texaco Regal Oil "C" at a speed of 500 r.p.m. Fig. 15 is for a load of 100 pounds per square inch; Fig. 16 is for 200 pounds; Fig. 17 shows 300 pounds, and Fig. 18 is for 400 pounds.

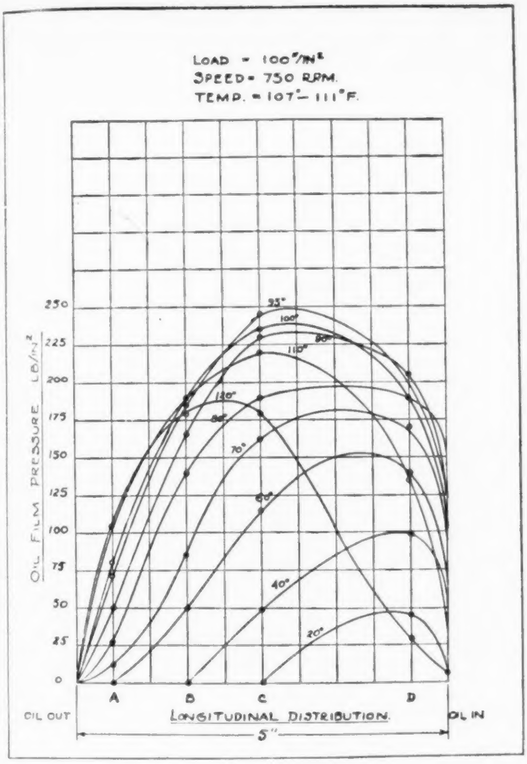

Figure 19

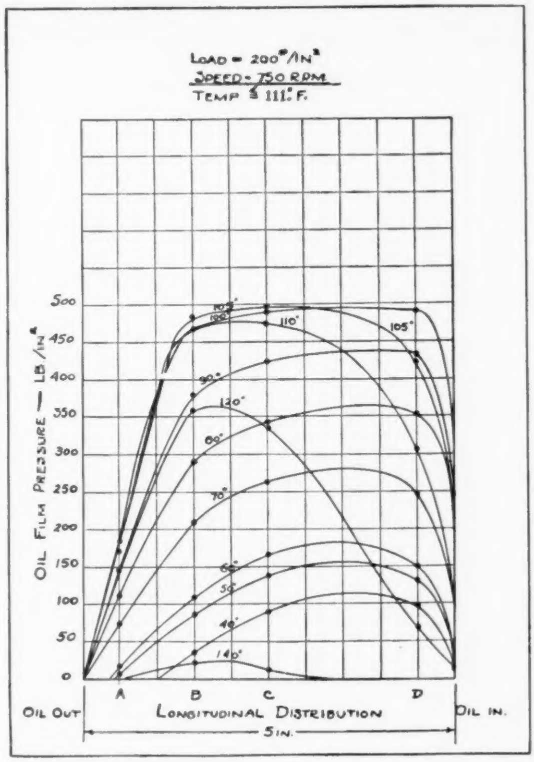

Figure 20

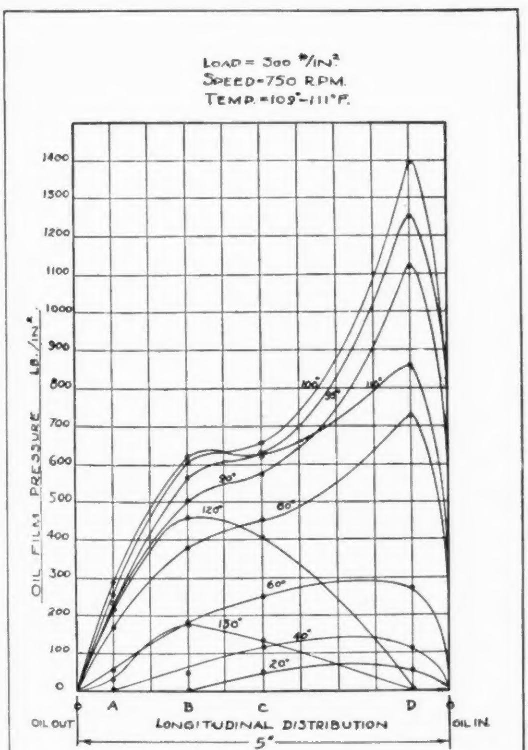

Figure 21

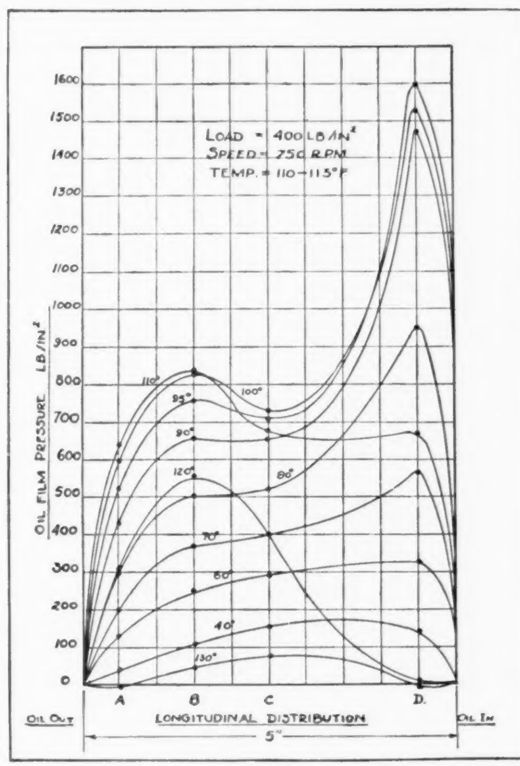

Figure 22

Showing longitudinal pressure curves for the same oil at a speed of 750 r.p.m., plotted in the same manner as shown on the preceding page. Fig. 19 shows a load of 100 pounds per square inch; Fig. 20 is for 200 pounds; Fig. 21 shows 300 pounds, and **Fig. 22 is for 400 pounds.**

CONCLUSION

The studies described in the foregoing pages are distinctive of the interest which is being shown in the application of the theory of lubrication to practical machine design and operation. It is indicative of the realization that effective bearing lubrication is an economic necessity, dictated by the present day high standards of engineering efficiency and competition. For proper lubrication, after all, means longer life, lower lubricant consumption, lower costs of maintenance, and repair, and higher efficiency in power transmission.

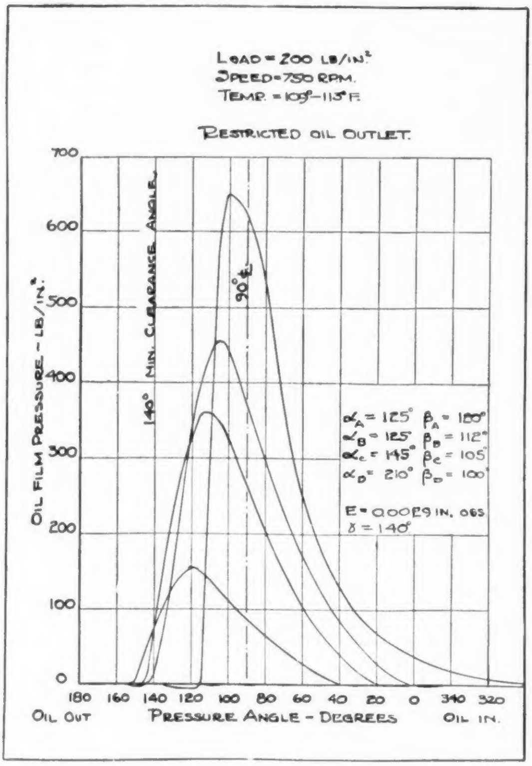

Figure 23

Showing pressure distribution with outlet oil sealed for operating conditions as indicated by the chart.

Studied in conjunction with such factors as load, operating speed, bearing clearance, means of application and the physical characteristics of lubricants, data as developed by the various authorities referred to in this article can be used to excellent advantage, if the practical side of the question is given due consideration. One must never lose sight of the fact, however, that the ultimate objection of the engineering personnel of any plant is to keep machinery running, to insure constant and maximum production, whatever the product. Research data should, therefore, be so developed as to further this program. It will be all the more appreciated by the practical operator in this regard and a source of satisfaction to the research student in the knowledge that his efforts have been directly applied to the furtherance of our worldwide needs for economical production.

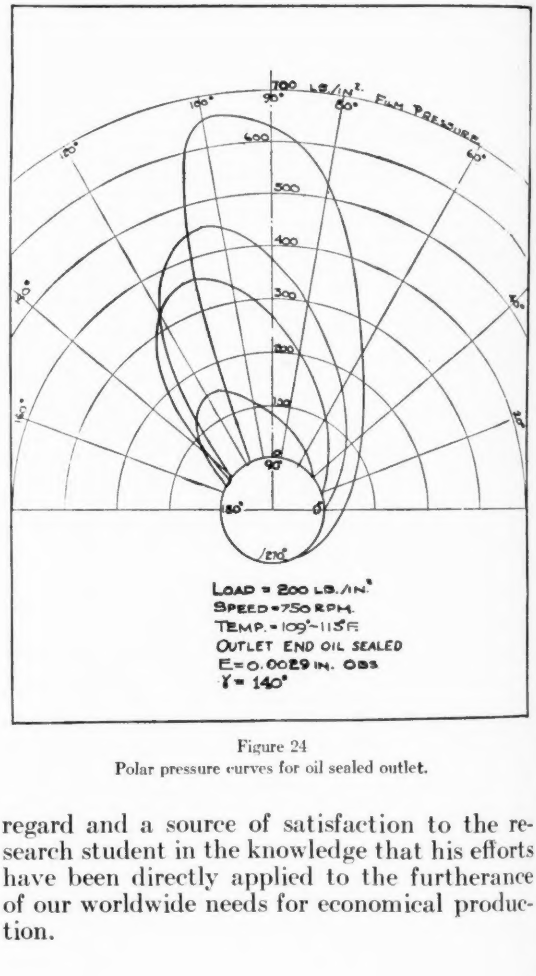

Figure 24

Polar pressure curves for oil sealed outlet.

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